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Calculation code for helically coiled heat recovery boilers

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Abstract

This paper describes a calculation code developed for helically coiled heat recovery boilers fed with exhaust gases from internal combustion engines, gas turbines or industrial processes. The code carries out the thermal rating calculation of the boiler by means of a one-dimensional model applicable to either water or thermal oil heating or steam generation in a once-through configuration. The paper is focused on the first case, in which cold fluid phase-change does not occur, and illustrates how the rating program can be helpful to improve the boiler design with respect to the current standard, allowing a reduction of several percentage points in the calculated heat transfer area (and correspondingly weight and material cost) required for a given duty.

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Keywords: Industrial energy efficiency; Waste heat; Helically coiled pipes; Heat recovery boilers; Heat exchangers; Thermal rating.

1. Introduction

Energy efficiency can limit demand growth, reduce energy imports and mitigate pollution. According to International Energy Agency, in the so-called New Policies Scenario, the IEA's scenario for energy outlook that considers existing policy commitments and those recently announced, efficiency accounts for about 70% of the reduction in projected global energy demand in 2035 if compared to the Current Policies Scenario, that assumes no implementation of policies beyond those adopted by mid-2012 [1]. One of the most effective strategies to improve energy efficiency in the industrial sector is waste heat recovery.

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Nomenclature

A	total heat transfer area on one side of the boiler, m^2
d	tube diameter (inside diameter for tube-side calculations, outside diameter for shell-side ones), m
D	coil mean diameter, m
h	specific enthalpy, J/kg
He	helical number (geometrical parameter for coils, dimensionless)
L	tube length, m
$\dot{m}$	fluid mass flow rate, kg/s
NC	number of coaxial coils
$NSEC$	number of computational sections
Nu	Nusselt number
p	coil pitch (that is the increase in coil elevation per revolution), m
Pr	Prandtl number
Re	Reynolds number
U	overall heat transfer coefficient, $W/m^2{\cdot}K$
u	average velocity inside tubes, m/s
Δh	generic specific enthalpy difference (outlet-inlet or inlet-outlet), J/kg
Δp	pressure drop, Pa
ΔT_{ml}	log-mean temperature difference, K, °C
Greek symbols	
η	dynamic viscosity, Pa·s
θ	angular location on the tube cross-section or helix angle of coil
λ	Darcy friction factor
ρ	fluid density, kg/m^3
Subscripts	
cr	critical
g	referred to gas stream
OUT	referred to outlet conditions
s	referred to the outside area of the coil
v	referred to cold fluid (water or thermal oil)
w	wall

It is estimated that waste heat ranges from 20 to 50% of industrial energy input in the form of hot exhaust gases, cooling water and heat losses from heated surfaces and products [2].

This paper describes a calculation code recently developed for helically coiled heat recovery boilers (water-tube type [3]) fed with exhaust gases from internal combustion engines, gas turbines or industrial processes. The code carries out the thermal rating calculation of the boiler by means of a one-dimensional model applicable either to water or thermal oil heating or steam generation in once-through configuration. The paper focuses on the first case, in which cold fluid phase-change does not occur, and, after a description of the code, shows that it is an effective design tool when a first geometrical configuration of the boiler is given. The code has been developed within a collaboration agreement with Garioni Naval SpA which deals with the design and manufacture of boilers and thermal machines.

2. Helically-coiled boiler description

Helically coiled tubes show some peculiar characteristics and phenomenological aspects of the thermo-hydraulics that are worthy of a brief description. First of all, coiled pipes are compact, can well accommodate the thermal expansions and have a high resistance to flow induced vibrations [4, 5]. Furthermore the fluid flowing in helical tubes develops secondary flows whose physical explanation is represented in Fig. 1. The curved shape of the tube causes the fluid to experience a centrifugal force which depends on the local axial velocity (Fig. 1 (a)). Due to the boundary layer, the fluid particles flowing close to the tube wall have a lower velocity with respect to the fluid flowing in the core of the tube thus they are subject to a lower centrifugal force [6, 7]. As a consequence, fluid from the core region is pushed outwards forming a pair of recirculating counter-rotating vortices (Fig. 1 (b)).

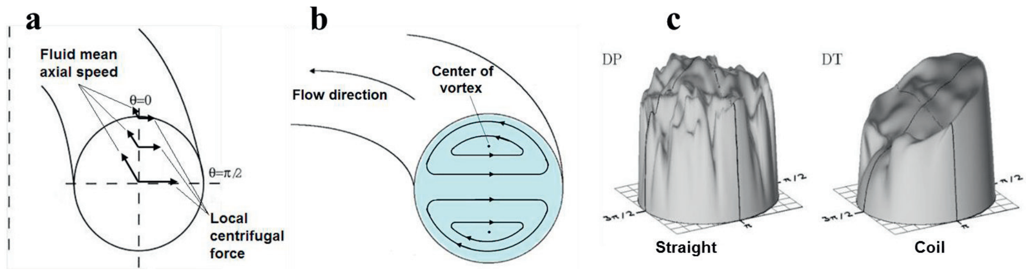

Fig. 1. (a) Centrifugal force acting on the flowing fluid and axial speed; (b) Resulting secondary flows; (c) Resulting profile of axial velocity [8].

The resulting profile of local axial velocity is different from that of straight pipes because the maximum axial velocity moves from the centre of the pipe to the outer wall ($\theta=\pi/2$, Fig. 1 (c)) [8]. Moreover the secondary flows produce an additional transport of fluid over the pipe cross section that increases both the heat transfer and the pressure drop of coils when compared to straight tubes [7]. The heat transfer rate is enhanced particularly in the laminar flow regime but slightly also at higher Reynolds number. Another effect of the development of secondary flows is represented by the more gradual and smooth transition from laminar to turbulent flow with respect to straight tubes [6].

The heat recovery boiler addressed by this analysis is a water-tube type [3] since the cold fluid passes through the pipes and the hot gases flow on the external side of the tubes. The tube bank for this kind of boiler consists of coaxial coils in the number of one up to six. Fig. 2 represents the longitudinal section of the boiler in case of two coils. As it can be seen in the drawing, the coils are inserted between two cylindrical shells, an external one, that is connected with two flanges, and an inner one that also acts as a by-pass duct since it is equipped with a throttle valve at one of its ends. At full load condition, the entire mass flow rate of gases passes between the two shells so that the heat exchange with the coils takes place, at partial load the control system partially opens the valve allowing a fraction of the gases to flow through the inner shell by-passing the coils.

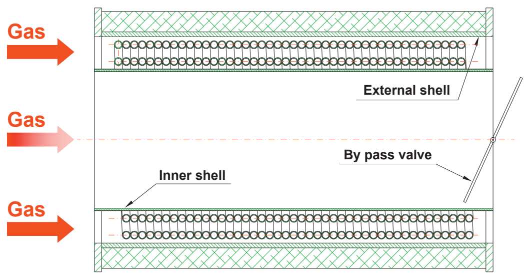

Fig. 2. Longitudinal section of the heat recovery boiler modeled by the code in case of two coils.

3. Mathematical model

The code carries out the thermal rating calculation of the boiler depicted above by means of a one-dimensional model. This paper focuses on the cases in which the boiler is employed for water or thermal oil heating. The rating process for boilers and heat exchangers runs the calculations of the thermal performance and of the pressure drops for a given geometry. The code developed in this work tackles the rating problem as it is at the basis of both the design and the verification procedures. In the design process, first a tentative set of geometrical parameters is selected and then the design is rated. On the other hand, when the geometry of a heat exchanger is already known, the performance analysis has to be conducted [9]. In both cases the rating calculations are needed. Moreover it has been considered that the rating procedure meets better the requirements of boilers manufacturers since they normally have preliminary geometrical configurations available. Fig. 3 gives the schematic representation of the rating program carried out by the calculation code. It receives as input data the mass flow rates and the properties of fluids, along with the inlet temperatures and pressures and the geometrical configuration of the boiler. By executing geometrical and heat transfer calculations, applying the conservation laws, employing the appropriate correlations, the code returns as output data the outlet temperatures and pressures and the thermal power transferred.

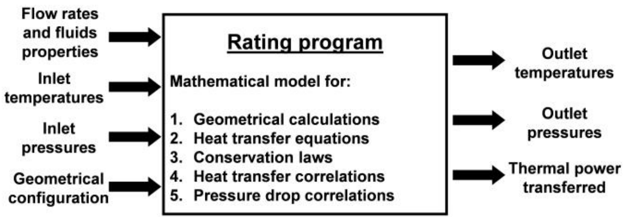

Fig. 3. Schematic representation of the rating program executed by the code.

The mathematical model subdivides the total gas flow rate in streams that flow between two consecutive coils or between the external coil and the outer shell or the internal coil and the inner shell. These streams can be fed in parallel arrangements or in series ones. The model considers separately each coil regard to the heat transfer calculation. The coils can be fed in series or in parallel or with intermediate solutions in parallel/in series. Furthermore the model allows to break the domain of calculation into longitudinal sections, primarily to take into account any geometrical irregularity, but also to subdivide the computational domain into multiple segments and thus perform a specific calculation on each portion. The possible geometrical discontinuities are: variations in the diameter and/or thickness of the tube, different axial lengths of the coils (winding length). In the longitudinal half-section of the boiler represented in Fig. 4 the subdivision of the computational domain into five sections as a consequence of geometrical discontinuities is shown.

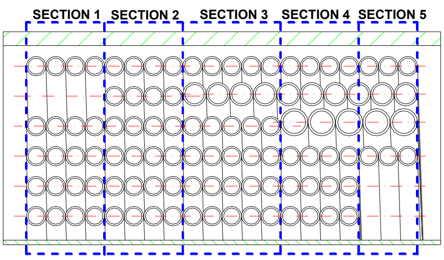

Fig. 4. Half-section of the boiler with subdivision of the calculation domain into five longitudinal sections.

All these options in terms of fluids feeding possibilities and of geometrical solutions have been included in order to give a high level of flexibility to the calculation code.

The modeling equations of the boiler are represented by the energy and mass balances and by the pressure drop calculation for each coil and each gas stream on every section. Since the overall inlet mass flow rates of the fluids and the inlet temperatures and pressures (Fig. 3) are given, the unknown variables are (where *NC* is the number of coaxial coils):

- *NC* outlet temperatures on the cold fluid side for every section.
- *NC+1* outlet temperatures on the gas side for every section.
- *NC* outlet pressures on the cold fluid side for every section.
- *NC+1* outlet pressures on the gas side for every section.
- *NC* mass flow rates (one for every coil) on the cold fluid side if the coils are fed in parallel.
- *NC+1* mass flow rates (one for every stream) on the gas side if the gas streams are fed in parallel.

Therefore, if *NSEC* is the number of computational sections, the resulting total number of unknowns is *(2NC+1)·(2NSEC+1)*. For instance, for the simple case of a boiler with 2 coils fed in parallel, gas streams in parallel and a unique section (Fig. 5 (b)), the total number of unknowns is equal to (2·2+1)·(2·1+1) = 15. In case of more complex configurations, namely with higher numbers of coils and computational sections, the number of unknown variables quickly increases. This remark suggests to split the overall system of equations into three sub-systems: the energy balance equations and the mass balances and pressure drop calculations on the cold fluid and gas sides.

3.1. Energy balance equations

The mathematical model is based on the hypothesis that the heat exchange occurs between the single gas stream and the coils that delimitate it. Each coil exposes half of its exchange area to each gas stream flowing on its outside. The gas streams flowing between the coil and the shells exchange heat only with one coil as the inner and the outer shells are assumed adiabatic. The heat transfer model can be depicted on a temperature-axial length diagram. Fig. 5 (a) shows such a diagram in case of two coils fed in parallel and the related three gas streams fed in parallel in a global counter-flow arrangement. The corresponding boiler configuration is reported as longitudinal half-section in Fig. 5 (b).

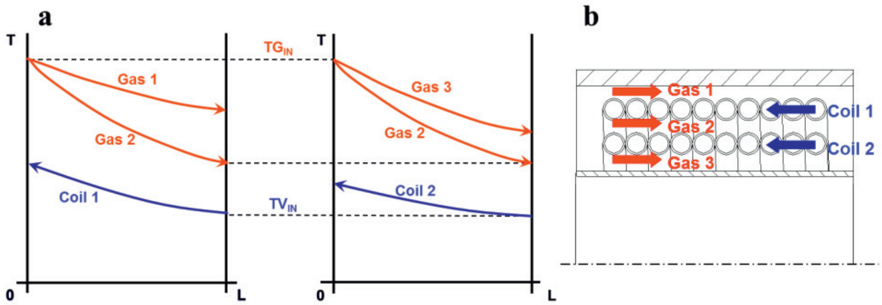

Fig. 5. (a) Temperature-axial length diagram for 2 coils, gas and coils in parallel, counter-flow ; (b) Half-section of the corresponding boiler.

The system of energy balance equations is attained by applying the conservation of energy to every coil and to every gas stream. Introducing the equation of overall heat exchange, the general structure of these relations is (Eq. 1):

$$\dot{m} \cdot \Delta h = UA\Delta T_{ml} \tag{1}$$

For the calculation of the heat transfer coefficients for the inside and the outside flows in helical pipes, and then for the evaluation of the overall heat transfer coefficient *U*, an in-depth literature review has been conducted and the resulting choices in terms of correlations are listed in Appendix A.

In this way, on every computational section, *NC+1* equations on the gas side and *NC* ones on the cold fluid side are attained, therefore the total number of equations is *NSEC·(2NC+1)*. Other equations to be considered are the congruence equations for temperatures between coils and/or gas streams fed in series. The unknowns are the fluids temperatures on the exit of every section, so there are *NSEC·(NC+1)* temperatures on the gas side and *NSEC·NC* ones on the cold fluid side, consequently the total number of unknowns is equal to *NSEC·(2NC+1)*. Other unknown variables to be considered are the entrance temperatures when coils and/or gas streams are fed in series. As a consequence, a non-linear (because of the logarithm in the expression to calculate the log-mean temperature) system of equations is obtained with the number of unknowns equal to the number of equations.

In order to better explain the energy balance equations, Eq. 2 represents the system written for a case with two coils, referring, for example, to the computational section labelled with index 1. The first equation refers to the gas stream 1: in the left hand side of the equation the mass flow rate is multiplied by the delta enthalpy, i.e. the difference between inlet enthalpy and outlet enthalpy for gas 1 on the section 1. In the right hand side of the equation: the overall heat transfer coefficient between the gas 1 and the coil 1 on the section 1 is multiplied by a half of the exchange surface of the coil 1 on the section 1 multiplied by the log-mean temperature difference between the gas 1 and the coil 1 on the section 1. And so on for other equations, this is the principle to write the whole system.

$$\begin{cases} \dot{m}_{g1}(h_{g1-1}-h_{g1-2}) = U_{1,1-1}\frac{1}{2}A_{s,1-1}\Delta T_{ml-1,1-1} \\ \dot{m}_{g2}(h_{g2-1}-h_{g2-2}) = U_{2,1-1}\frac{1}{2}A_{s,1-1}\Delta T_{ml-2,1-1} + U_{2,2-1}\frac{1}{2}A_{s,2-1}\Delta T_{ml-2,2-1} \\ \dot{m}_{g3}(h_{g3,1}-h_{g3,2}) = U_{3,2-1}\frac{1}{2}A_{s,2-1}\Delta T_{ml-3,2-1} \\ \dot{m}_{v1}(h_{v1,2}-h_{v1,1}) = U_{1,1-1}\frac{1}{2}A_{s,1-1}\Delta T_{ml-1,1-1} + U_{2,1-1}\frac{1}{2}A_{s,1-1}\Delta T_{ml-2,1-1} \\ \dot{m}_{v2}(h_{v2,2}-h_{v2,1}) = U_{2,2-1}\frac{1}{2}A_{s,2-1}\Delta T_{ml-2,2-1} + U_{3,2-1}\frac{1}{2}A_{s,2-1}\Delta T_{ml-3,2-1} \end{cases} \tag{2}$$

To clarify the use of the subscripts, some examples are given: A_{s1-2} means outside area of coil 1 on computational section 2; h_{g1-2} means enthalpy of gas stream 1 at point 2 (extreme point of a computational section); h_{v1-3} indicates enthalpy of cold fluid in coil 1 at point 3 (extreme point of a computational section); $U_{3,2-1}$ is the overall coefficient for gas stream 3 and coil 2 on computational section 1; $\Delta T_{ml,3,2-1}$ represents the log-mean temperature for gas stream 3 and coil 2 on computational section 1.

3.2. Mass balance and pressure losses equations – cold fluid side

The mathematical model applies the equation of conservation of mass to calculate how the total cold fluid flow rate is split into the coils in case of parallel feeding (Eq. 3):

$$\dot{m}_v = \dot{m}_{v1} + \dot{m}_{v2} + \ldots\ldots + \dot{m}_{vNC} \tag{3}$$

Moreover the model applies the momentum equation to every coil in order to evaluate on each computational section the pressure drop and therefore the entrance and exit pressure values on the sections. Eq. 4, where λ is the Darcy friction factor, gives the general expression for calculating the pressure losses:

$$\Delta p = \lambda \cdot \rho \cdot \frac{u^2}{2} \cdot \frac{L}{d} \tag{4}$$

Suitable correlations for determining the friction factor in case of inside flow in helically-coiled tubes are presented in Appendix A as a result of a wide investigation of literature conducted in this work.

For coils fed in parallel the system includes also the relations between the exit pressures of each coil since these values must be equal each other to prevent the recirculation of the fluid (Eq. 5):

$$p_{v1,OUT} = p_{v2,OUT} = \ldots = p_{vNC,OUT} \tag{5}$$

Finally, if all coils are fed in parallel, the system of equations to be solved is:

- 1 equation for the mass balance.
- *NSEC·NC* equations to calculate the pressure loss on every section.
- *NC-1* relations between the exit pressure of each coils.

Therefore *NC·(NSEC+1)* total equations are written. The unknown variables are: *NC* flow rates (one for every coil) and *NSEC·NC* exit pressure for each coil, hence the total number of unknowns is *NC·(NSEC+1).* In case of coils fed in series, in addition there are *NC-1* congruence equations between the extreme pressure values of each coil but the mass balance is no longer needed because the flow rate is unique and known. *NC-1* additional unknowns appear because of the unknown entrance pressure values of coils fed in series. On the cold fluid side the code allows to set intermediate solutions for the coils feeding, namely "hybrid" configurations in series/in parallel. In any case the mathematical model gives rise to a non-linear (because of the fluid velocity squared) system of equations with number of equations equal to the number of unknowns, even in the special cases in which the complexity of the model increases considerably (different coils axial lengths, hybrid feeding, extreme coils in contact with shells).

3.3. Mass balance and pressure losses equations – gas side

With the same approach the model deals with the mass balance and the pressure losses on gas side. Appropriate correlations to evaluate the friction factor on the shell-side of helical tubes bundles are presented in Appendix A.

In case of parallel feeding of the gas streams the subsequent equations have to be considered:

- 1 equation for the mass balance.
- *NSEC·(NC+1)* equations to calculate the pressure loss on every section.
- *NC* relations between the exit pressure of each gas stream: these values must be equal to each other to prevent the recirculation of gas.

Thus *(NC+1)·(NSEC+1)* total equations are written. The unknown variables are: *NC+1* flow rates (one for every gas stream) and *NSEC·(NC+1)* exit pressure for the gas streams, then the total number of unknowns is equal to *(NC+1)·(NSEC+1).* In every case, included that of streams fed in series, a non-linear system of equations with the number of equations equal to the number of unknown variables arises from the previous considerations.

3.4. Numerical resolution of the mathematical model

As previously stated, the model gives rise to a high number of non-linear equations, hence the code divides the overall system of equations into the three sub-systems above described. These systems are solved separately from each other with a modified Powell hybrid algorithm [10] which proceeds by subsequent iterations, starting from an initial estimate of the roots. The three systems are connected together since they have common variables (e.g. the mass flow rates are determined by the mass balances and by the relations between the pressures and are used in the energy balance). For this reason, the application of the Powell algorithm to each system occurs in the framework of an overall cycle that is repeated until the convergence is reached, namely when the deviation between the results of two subsequent iterations is lower than a threshold fixed.

4. Calculations results

As previously highlighted, a rating program is useful to refine a preliminary design. For example, the code developed in this work has been used to improve the standard design of two models of thermal oil boiler. Since it has been noticed that in the basic design cases the outlet temperatures of the gas streams and of the cold fluid flow rates are not uniform and this causes a decrease of the boiler efficiency, the geometry has been improved in order to achieve more uniform outlet temperatures on the gas side. In particular, to reach this goal, the coils winding diameters and the shells diameters have been modified, as it results in Table 1, aiming at a different distribution of the gas mass flow rate into the streams flowing across the coils. Afterwards the axial length of the boiler has been reduced to have the same value of the heat transferred that characterizes the base case. In this way a comparison between the basic design and the improved one for a given duty has been performed. The results, represented in Table 1 and in Fig. 6, point out the reduction in terms of weight (of tubes and shells, because of the lower heat transfer area required) of the boiler that is achieved by refining the current standard design for the same heat duty. The weight of the boiler lower by several percentage points corresponds to a minor material cost. This kind of modification has been presented as an example, but the code allows more complex variations of the geometry of the boiler to refine the design.

Table 1. Results of the code calculations in terms of boilers weight for basic and improved design for two boilers.

Case	Option	Boiler Heat Duty [kW]	Diam. External Shell [mm]	Coil Winding Diameter [mm]				Diam. Internal Shell [mm]	Axial Length [mm]	Pipe Total Length [m]	Total Weight [kg]	Weight Reduction [%]
				Coil 1	Coil 2	Coil 3	Coil 4					
Case 1	Basic	504,67	1790	1650	1500	1370	1245	1202	1223,80	540,63	2374,54	6,62
	Impr.	504,67	1770	1650	1510	1370	1225	1180	1148,00	506,29	2217,43	
Case 2	Basic	772,78	1900	1740	1570	1420	1245	1206	1738,80	697,64	3469,90	4,93
	Impr.	772,58	1878	1738	1572	1408	1242	1190	1660,00	664,41	3298,74	

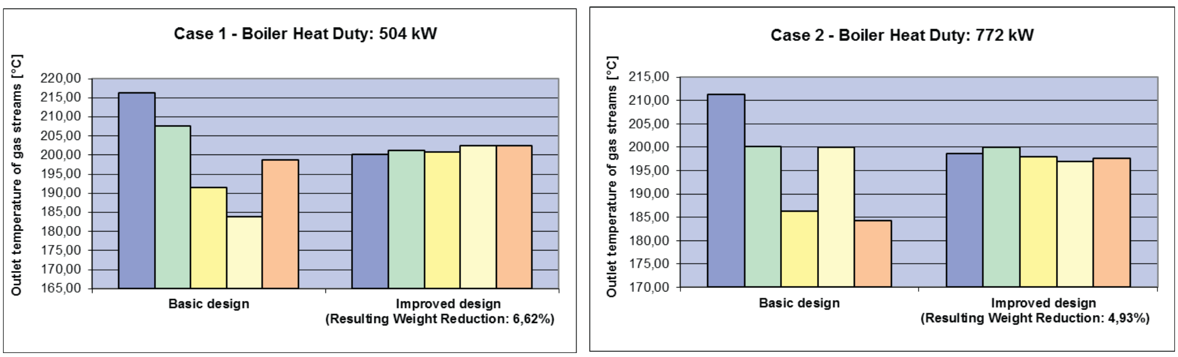

Fig. 6. Distribution of the outlet temperatures of the gas streams in case of basic and improved design for two boilers.

5. Conclusions and further development

The results exposed above demonstrate that the code developed for the thermal rating of helically coiled heat recovery boilers within a collaboration agreement with Garioni Naval is an effective design tool when a preliminary geometrical configuration is given. The code is characterized by a high level of flexibility since it allows to set many arrangements in terms of fluids feeding combinations and of geometrical solutions for the tube bundle, moreover the model implemented offers different options as regards the correlations and the methods to evaluate pressure drop and heat transfer on both the tube-side and the shell-side.

The code could be validated by an experimental activity to verify the mathematical model developed (for example in relation also to the uncertainties introduced by the heat transfer correlations). The code is already applicable to heat recovery once-through steam generators with helical pipe banks therefore a further development of this work concerns the description of this part of the model.

Acknowledgements

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Appendix A. Pressure losses and heat transfer correlations employed in the code

A.1. Tube-side, pressure losses

The Ito's correlation [6, 11] for turbulent flow ($Re>Re_{cr}$, Eq. 6) is:

$$\lambda = 0{,}304 \cdot \mathrm{Re}^{-0{,}25} + 0{,}029 \cdot \left(\frac{D}{d}\right)^{-0{,}5} \text{ multiplied by the correction term } \left(\frac{\mathrm{Pr}}{\mathrm{Pr}_w}\right)^{-0{,}33} \tag{6}$$

$$\mathrm{Re}_{cr} = 2100 \cdot \left[1 + 12 \cdot \left(\frac{D}{d}\right)^{-0{,}5}\right] \text{ expression to calculate the critical Reynolds number [12]} \tag{7}$$

The Mishra and Gupta's correlation [13, 14] for turbulent flow ($Re>Re_{cr}$, Eq. 8) is:

$$\lambda = 0{,}3164 \cdot \mathrm{Re}^{-0{,}25} + 0{,}03 \cdot \left(\frac{D}{d}\right)^{-0{,}5} \tag{8}$$

A.2. Tube-side, heat transfer

Gnielinski's correlation [7, 14] for laminar flow ($Re<Re_{cr}$, Eq. 9) and Schmidt's [14] correlation for turbulent flow ($Re>2{,}2\cdot 10^4$, Eq. 10) are:

$$Nu = 3{,}65 + 0{,}08 \cdot \left[1 + 0{,}8 \cdot \left(\frac{d}{D}\right)^{0{,}9}\right] \cdot \mathrm{Re}^m \cdot \mathrm{Pr}^{1/3} \cdot \left(\frac{\mathrm{Pr}}{\mathrm{Pr}_w}\right)^{0{,}14} \text{ with } m = 0{,}5 + 0{,}2903 \cdot \left(\frac{d}{D}\right)^{0{,}194} \tag{9}$$

$$Nu = \frac{(\lambda/8) \cdot \mathrm{Re} \cdot \mathrm{Pr}}{1 + 12{,}7 \cdot \sqrt{\lambda/8} \cdot \left(\mathrm{Pr}^{2/3} - 1\right)} \cdot \left(\frac{\mathrm{Pr}}{\mathrm{Pr}_w}\right)^{0{,}14} \tag{10}$$

Where λ is the friction factor for turbulent flow given by Mishra and Gupta [13] multiplied by the correction term $(\eta_w/\eta)^{0,27}$ as ratio between the dynamic viscosity of the fluid at the wall and bulk temperature. For the transition region ($Re_{cr} <Re<2{,}2\cdot 10^4$) the code applies a linear interpolation between Gnielinski and Schmidt correlations as suggested in [14].

Mori-Nakayama's correlation [4] for laminar and turbulent flow ($1000<Re<100000$, Eq. 11) is:

$$Nu = \frac{\mathrm{Pr}^{0,4} \cdot \mathrm{Re}^{5/6} \cdot (d/D)^{1/12} \cdot \left[1 + \frac{0,061}{\left[\mathrm{Re} \cdot (d/D)^{2,5}\right]^{0,167}} \right]}{41} \tag{11}$$

A.3. Shell-side, pressure losses

Le Feuvre method [4, 15] is based on the conclusion that cross-inclined banks of straight tubes can be used to model helically-coiled heat exchangers. The calculation procedure of the overall pressure drop across the helical tubes bundle is presented in [15].

A.4. Shell-side, heat transfer

The correlation proposed by Zukauskas for straight tube bundles in-line is suitable also for helical coils [4, 12] in the range 1000<*Re*<200000 and for more than 16 rows. *Re* calculation is based on tube outside diameter *d*:

$$Nu = 0,27 \cdot \mathrm{Re}^{0,63} \cdot \mathrm{Pr}^{0,36} \cdot (\mathrm{Pr}/\mathrm{Pr}_w)^{0,25} \tag{12}$$

The correlations recommended by Abazdic [4], with *Re* based on tube outside diameter *d*, are:

$$Nu = 0,332 \cdot \mathrm{Re}^{0,6} \cdot \mathrm{Pr}^{0,36} \text{ for } 1000<Re<20000 \tag{13}$$

$$Nu = 0,123 \cdot \mathrm{Re}^{0,7} \cdot \mathrm{Pr}^{0,36} \text{ for } 20000<Re<200000 \tag{14}$$

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